Design and Analysis of the Drive Train System of an All-Terrain Vehicle.

Submitted in partial fulfillment of requirements

For the degree of

Bachelors in Mechanical Engineering

by

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Certificate

This is to certify that the dissertation entitled "Design and Analysis of the Drive Train System of an All-Terrain Vehicle" is bona fide record of the dissertation work done by

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in the year 2014-15 under the guidance of **V.S. Narwane** of Department of Mechanical Engineering in partial fulfillment of requirement for the Bachelors Degree in Mechanical Engineering of University of Mumbai.

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Certificate of Approval of Examiners

We certify that this project report entitled "Design and Analysis of the Drive Train System of an All-Terrain Vehicle" is bona fide record of project work done by

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Abstract

Drivetrain basically concerns with conveying power from the vehicle's engine, through the transmission to the drive wheels on the vehicle to control the amount of torque. It comprises of the Engine, CVT (Continuously Variable Transmission), Gearbox, Differential, CV joints and Axle shafts. The ATVs are meant for operation on rough, off-road terrains which demand higher torque and a smooth transmission. The BAJA competition allows use of a standard 10Hp engine from which power transmission is to occur with appropriate torque conversion. An automatic transmission system through the use of a CVT has been incorporated due to its advantage of having smooth operation (infinite gear ratios), elimination of clutch, gear shifting and compactness over manual transmission. Also CVT tuning enables us to operate the engine at power peak so that power is transmitted most efficiently. A self-designed two stage reduction gearbox is coupled with the CVT to achieve the desired torques requirements for the terrain. An open differential is integrated with the gear train to provide aid in steering. The motion is further transferred from the gearbox to the wheel assembly through CV joints because of their ability to transmit torque through a higher range of suspension articulation. While designing and analysis emphasis is laid on, material selection, selection of proper FOS, method of production for gears and the gearbox so as to ensure longer life for the system, minimal transmission losses and also minimize the total cost of the system so that it can compete in the existing market. Also proper system layout has been ensured for ease in serviceability. These features will create a vehicle that utilizes all of its power in a smooth, quick transition from rest to top speed, and insures minimal maintenance.

Keywords: Drive Train, Engine, Continuously Variable Transmission, Gearbox, Differential, CV joint, Torque, Serviceability

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Chapter 1

Introduction

This chapter presents a brief introduction involving the background and motivation behind the project and the proposed research work undertaken in order to achieve the desired objective.

1.1 Background

The objective of the competition is to provide students with a challenging project that involves design, engineering, planning, manufacturing and marketing the tasks found when introducing a new product to the consumer industrial market. Teams compete against one another to have their design accepted for manufacture by a fictitious firm. Students must function as a team not only to design, build, test, promote, and run a vehicle within the limits of the rules, but also to generate financial support for their project and manage their educational priorities.

Our project focuses on the study, design, analysis and manufacture of the power transmission system of the BAJA all-terrain vehicle in association with team RedShift Racing India. The project encompasses the study of the previous gearbox designs of the team, the Continuously Variable Transmission System (CVT), and the development of a new design for a better performance of the vehicle.

We focus on the optimization of the current design in terms of the cost of manufacture and the overall weight of the system, along with the improvement in the performance characteristics like acceleration, maneuverability and speed. We design the components on CATIA software while carrying out structural analysis on ANSYS software. The project involves the design of the entire two stage reduction gearbox, study of the performance characteristics of the CVT and its suitable modification with rigorous testing, design and manufacture of the rear-axle, and stability and mounting considerations of the entire system in particular.

1.2 Proposed Research Work

Our current research work has three main subdivisions:

1. Design and analysis of components including the gears, rear axle, housing, etc.
2. Study of performance characteristics of the CVT and parameters that determine them. Study of CVT tuning and implementation.
3. Study of Vibrational analysis of an engine. Experimental analysis of the current engine and suitable damping and mounting of the engine.

<u>For the reduction gearbox:</u>

- Gear design and material selection. Gear manufacturing processes, their reliability, cost optimization. Hardening processes.

- Housing design. Materials properties required for sand casting. Design considerations for sand casting. Determination of tolerances for bearing seats, shaft designs. CNC machining for critical sections.
- Bearing and oil seals selection.
- FEA method for structural analysis.

For CVT systems:

- Understanding the working of the CVT systems. Forces and basic physics of the working.
- Study and determination of working parameters such as engagement RPM, shift RPM, etc.
- Components determining parameters for design such as properties of springs, torsional springs, flyweight design and manufacture, ramp angles for shift, etc
- Modification of existing system through reverse engineering for suitable performance.

Engine Mounting:

- Study of Vibrational analysis and vibration damping.
- Experimentation for vibration characteristics of the engine.
- Selection of suitable damping supports for reduction in transmitted vibrations.

According to our market research, the proposed aim of the project can be achieved satisfactorily. Following are some limitations for the scope of the project:

- Pressure Die casting process is very costly for single product manufacture. Therefore, housing of the prototype will be manufactured using simple sand casting.
- Accurate vibrational analysis and exact material specifications need acute measurement and resources. The project will therefore look towards optimization with available resources.

1.3 Motivation

The BAJA competition requires a robust and high capacity transmission system to ensure the expected level of performance in a rough terrain. The need for better performance and higher ranks in the competition in comparison with previous years compels us to continually modify and improve the existing transmission system.

Chapter 2

Literature Survey

This chapter presents a review of the references and theories related to the project that were adopted as basis for designs and calculations. It gives a brief summary for the reader to understand the background of our design and implementation.

2.1 Continuously Variable Transmission

The CVT system consists of three components. The driver pulley, the driven pulley and the driving belt. The driven clutch consists of two stamped steel sheaves. One is a fixed sheave while the other is a movable one, with a pressure spring pushed against it. A centrifugal mechanism and along with the pressure springs controls the movement of the sheaves which varies the velocity ratio of the drivetrain. [1] Given below are the two main aspects of CVT tuning.

Engagement speed:
The engagement speed is the RPM of the driver of the CVT at which the belt gets engaged with the sheaves to rotate the driven. It is determined by the driver pressure spring and the flyweights. Since the centrifugal force is directly proportional to the RPM, above a particular RPM, the centrifugal force of the flyweights just overpowers the pressure spring force opposing it. At this stage, the sheave moves inwards gripping the belt, and hence engaging the driven clutch. The engagement RPM is selected close to the maximum torque peak of the engine, so that maximum torque output is achieved at low gear ratio.

Shift speed:
The shift speed is the speed of the engine where the velocity ratio shifts from the lower ratio to the higher ratio. It is the RPM at which the sheave on the driver pulley moves further inside and the sheave on the driven pulley moves outwards so as to change the velocity ratio of the system. It is determined by the ramp angles provided for the sliding of the sheave, and the pretension provided on the torsional spring in the driven clutch. At shift speed, the centrifugal force of the flyweights exceeds the spring force in the driver clutch as well as the side force on the belt at the driven clutch due to the ramp angle and the torsion spring. Hence the sheave on the driver clutch starts moving further inwards till it reaches its maximum limit at the higher gear ratio.

The CVT should be tuned such that the engine runs in the power peak range under its entire range of operation. This can be achieved by varying the parameters like the spring constant, spring pretension, torsional stiffness of the spring, flyweight masses and ramp angle.

2.2 Gearbox Design

Ratio and Gear Selection:

For a reduction ratio of above 4, a multiple stage compound gear train is recommended. A two stage drive train is suited for the reduction ratio of about 14, with about 3.7 reduction ratio at each stage. Helical gears perform well in high speed operations where smooth transmission and low noise is essential. Helical gears are therefore suited for vehicle gear box systems. Full depth involute system is the most commonly used system for the gear profile suited for high speed transmissions.

Module and FOS:

Optimum number of teeth are selected on the basis of minimum weight and elimination of interference phenomenon. Modules are selected on the basis of bending strength and the wear strength of the tooth profiles. Lewis equation is adopted for the calculation of the bending and wear strengths of the gears. Uniform teeth profile, load application on a single tooth at a time, neglecting the radial and compressive forces, are some of the assumptions made during the calculations. To account for the neglected factors, a suitable factor of safety of 2.5 is selected for the selection of the modules.

Shafts and Keys:

Shaft design is based on the torsional equations. Length of shaft is determined considering limited elastic deformation during operation. Keys are designed considering shear and crushing failure. Standard sizes are adopted to approximate dimensions to the nearest values.

Bearings:

Bearings are selected based on the load ratings provided in the SKF bearing catalogue. Selection of bearings is based on the factors like axial loads, radial loads, average speed of operation and life of the bearing. Tapered roller bearings are suited in operations where a slight axial offset or angle, and where axial force is considerable. Tapered roller bearings also aid the ease of assembly of components. They are therefore suited in applications where there may be a slight error in manufacture of housings, and where frequent assembly and disassembly is required. Hence, they are selected over the other types of bearings available.

Housing Design:

The housing design must be able to accommodate all the gear and shaft components within a compact space avoiding interference at the same time. Its design should also be such that assembly requires minimum time and is easy. It constitutes space for the gears, and seats for the bearings. The casing should sustain forces like weight of the gears, radial and axial reactions of the bearings, and any outside impact to protect the components from damage. The housing should also prevent leakage of lubricating oil. Housing design is based on basic principles of Machine Design and accepted theories of failure in tensile, bending and shear modes. Limited elastic deformation is also considered while design. Owing to the complex nature of the design, selection of one factor of safety for all design features is redundant. The design is thus first made based on judgement, analysed on a computational analysis software and then suitable changes are made towards optimality.

2.3 Engine Mounting and Vibration Analysis

Vibrations:

Vibration is a response of a system subjected to disturbance. It is inter-conversion of kinetic energy into potential energy and vice versa. Vibrations are oscillations of a mechanical or structural system about an equilibrium position. They are initiated when an inertia element is displaced from its equilibrium position due to an energy imparted to the system through an external source. Any inertia element can undergo vibrations and has its own natural frequency.

Effects of Vibrations:

Vibrations occur in many mechanical and structural systems. If uncontrolled, vibration can lead to catastrophic situations.

1. Vibrations in mechanical systems results in loosening of parts and can in general create a hazardous situation.
2. It leads to rapid wear of machine parts such as bearings, gears etc. because of which their life decreases appreciably.
3. Vibrations of machine tools or machine tool chatter can lead to improper machining of parts.
4. Structural failure can occur because of large dynamic stresses developed during earthquakes or even wind-induced vibration.
5. Excessive vibrations of pumps, compressors, turbo-machinery, and other industrial machines can induce vibrations of the surrounding structure, leading to inefficient operation of the machines while the noise produced can cause human discomfort.

Thus keeping in view all this devastating factors, the study of vibration is essential to minimize the vibrational effects over the mechanical components by designing them suitably.

Vibrations in IC Engines:

Vibrations occur wherever there is rotating machinery or moving parts. In IC Engine, the output shaft rotates at high speeds. Due to the manufacturing error in the shaft the centre of gravity of the shaft does not lie on the axis of rotation but is slightly eccentric because of which centrifugal force acts when the shaft rotates.

As the centrifugal force is directly proportional to the square of the speed of rotation therefore when shaft rotates at high speeds, the high amount of centrifugal force generated causes the engine to vibrate.

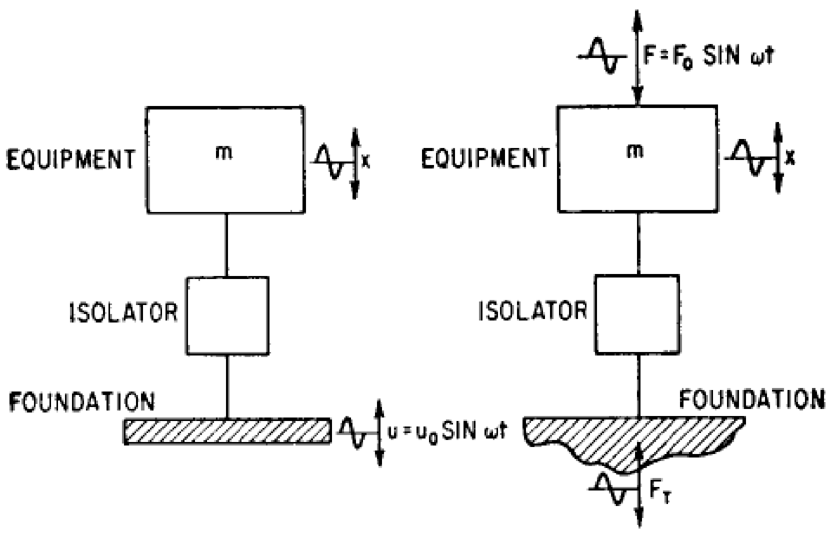
This force is called excitation force acting on the engine which is responsible for vibrations and the speed at which this excitation force acts is called external excitation frequency.

Resonance in the system occurs if the external excitation frequency is equal to the natural frequency of the system. It is absolutely necessary to ensure that the speed of the shaft is nowhere near the resonance frequencies of the system, as that would mean failure of the line shaft and components of the engine. Thus determining the natural frequency of the system is most important in the study of vibrations.

Due to the external excitation force acting on the engine, force is transmitted to its mounts. The ratio of the transmitted force to excitation force is called transmissibility ratio. Proper concepts of forced vibrations are to be applied in order to keep this ratio less than one i.e. to keep the system in isolation zone.

Vibration Isolation:

Vibration isolation means to bring about a reduction in a vibratory effect. A vibration isolator in its most elementary form may be considered as a resilient member connecting the equipment and foundation. The function of an isolator is to reduce the magnitude of motion transmitted from a vibrating foundation to the equipment or to reduce the magnitude of force transmitted from the equipment to its foundation.



The concept of vibration isolation is illustrated by consideration of the single degree of freedom system illustrated in the figure below. This system consists of a rigid body representing an equipment connected to a foundation by an isolator having resilience and energy dissipating characteristics. It is assumed that the body is constrained to move only in vertical direction.

The performance of the isolator is determined by its transmissibility ratio (T). Transmissibility is a measure of the reduction of transmitted force or motion afforded by an isolator. If the source of vibration is an oscillating motion of the foundation (motion excitation), transmissibility is the ratio of the vibration amplitude of the equipment to the vibration amplitude of the foundation. If the source of vibration is an oscillating force originating within the equipment (force excitation), transmissibility is the ratio of the force amplitude transmitted to the foundation to the amplitude of the exciting force.

Types of isolators:

The essential features of an isolator are resilient load-supporting means and energy dissipating means. In certain types of isolators, the functions of the load-supporting means and the energy-dissipating means may be performed by a single element, e.g., natural or synthetic rubber. In other types of isolators, the resilient load-carrying means may lack sufficient energy-dissipating characteristics, e.g., metal springs; in which case separate and distinct energy-dissipating means (dampers) are provided.

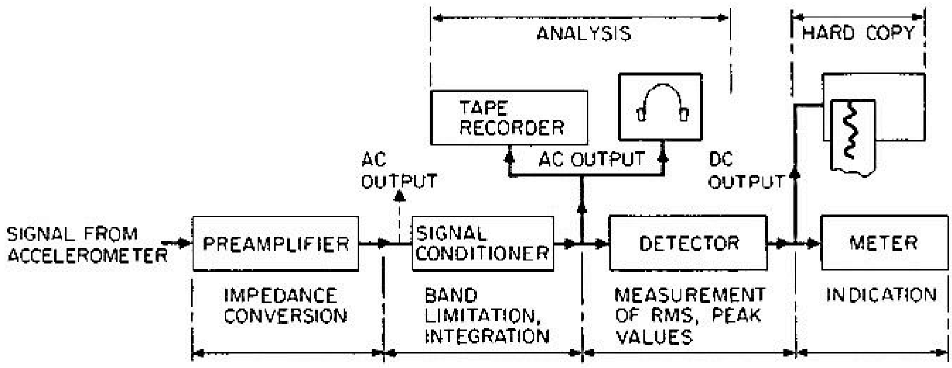
Rigidly connected viscous damper: A viscous damper c is connected rigidly between the equipment and its foundation as shown in Table 30.1A. The damper has the characteristic property of transmitting a force F_c that is directly proportional to the relative velocity $\dot{\delta}$ across the damper, $F_c = c\dot{\delta}$. This damper sometimes is referred to as a linear damper.

Measuring Instruments:

Vibration transducer:

The figure below shows a typical measurement system consisting of a preamplifier, a signal conditioner, a detector, and an indicating meter. Most or all of these elements often are combined into a single unit called a vibration meter.

Vibration meters are instruments which receive a signal from a vibration transducer and process it so as to give an indication of relevant vibration parameters. For measurements on most rotating machines, a frequency range of 10 Hz to 10 kHz is desirable. The lower limit includes the shaft speed for all machines operating over 600 rpm.



Vibration meter known as vibrometer gives us peak to peak amplitude (displacement) of vibration in microns. Knowing the displacement velocity and acceleration can be easily calculated.

2.4 Material Selection

Materials are selected based on their properties, availability, cost and suitability for the required application. Some examples of properties considered for material selection are yield strength, ultimate tensile strength, hardness, ability for heat treatment, machinability, allowable working temperature.

The materials for gears are selected on the basis of tensile strength, compressive strength and suitability for surface hardening. Mild steel of grade EN24-EN36 are suitable for these applications.

The housing, which is to be manufactured by casting process demands a material which has high flow ability, rigidity, less porosity after casting, high tensile and compressive strength, good corrosion resistance. Low weight is also a dominant factor in the design of the housing. The LM series of aluminium is suited for sand casting. It is also light in weight. Considering optimization of cost, LM6 grade aluminium is used for casting.

2.5 Objective of the Project

The project objective is to design a smooth and efficient transmission system to successfully complete the ‘Hill Climb’, ‘Traction’ and ‘Endurance’ events at the BAJA SAE competition. The team aims at providing maximum acceleration through CVT tuning to secure a better place in the acceleration event. In a general perspective, the objective of the project is to design an optimized transmission system to match and compete with the systems present in current ATVs with a lower manufacturing price.

Chapter 3
Elements of the Drivetrain

This chapter presents detailed information about the components used in the powertrain of the ATV. It gives the description of the elements involved and their functions in the working of the drivetrain.

3.1 The Engine

The BAJA SAE All-Terrain Vehicle uses a single cylinder, over-head valve, petrol engine with a capacity of 4.5 litres. The picture below shows the front view and side view of the current engine.

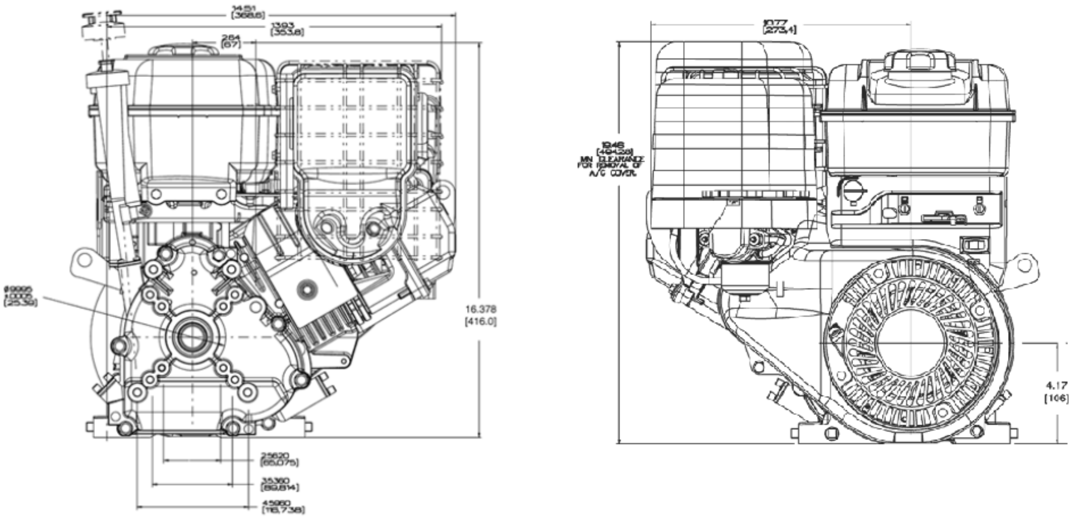


Fig. 1.1

The engine used is a Briggs and Stratton Intek Series four-stroke 10HP 305cc air-cooled engine having a peak torque output of about 14.5 lb-ft. According to the competition specifications, the engine runs in the range of 1750 RPM (idle) to 3800 RPM. The weight of the engine is 23 Kgs.

Although the engine is not a part of the drive train, study and tuning of the engine is important for the effective design of the drive train. Damping of the vibrations generated by the engine is necessary to avoid damage to the drive train components and reduce losses. A detailed experimentation on this topic is carried out in chapter 9.

To study the characteristics of the engine, i.e. torque and power curves, dyno-testing of the engine has been carried out. The following graph shows the results of the dyno-test carried out on the engine within working range.

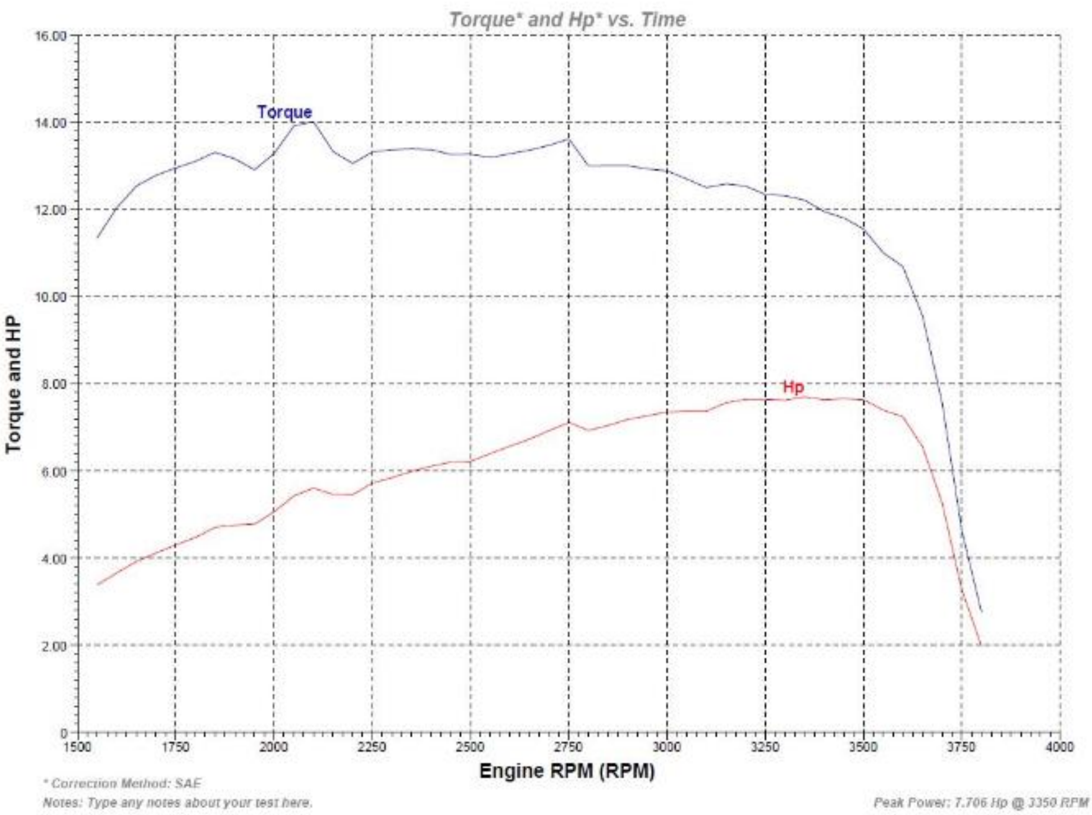


Fig. 1.2

The important results obtained from the dyno-test are as follows:

1. Peak torque output: 14.1 lb-ft.
2. RPM at peak torque: 2100-2200 RPM
3. Peak power output: 7.9 HP
4. RPM at peak power output: 3400-3500 RPM

These parameters are important for the design of the drive train.

3.2 Continuously Variable Transmission system (CVT)

A continuously variable transmission system is a ‘smooth uninterrupted power delivery system automatically adjusting itself to any load conditions’. CVT systems have gained popularity in the recent years, owing to the fact that they are compact, simple in construction and modification and are more efficient than traditional transmission systems.

The basic structure of a belt driven CVT includes two cone clutches, each having one fixed, and one movable sheave. The clutch connected to the engine is called the driver clutch, while the clutch at the output is called the driven clutch. At the driver clutch, the movable sheave is pushed away from the belt by a pressure spring, and a centrifugal mechanism with flyweights works against it. When the centrifugal force developed is more than the spring force, the sheave moves towards the belt and grips the belt to engage the transmission system.

The following diagram shows a cross sectional view of a simple belt driven CVT mechanism.

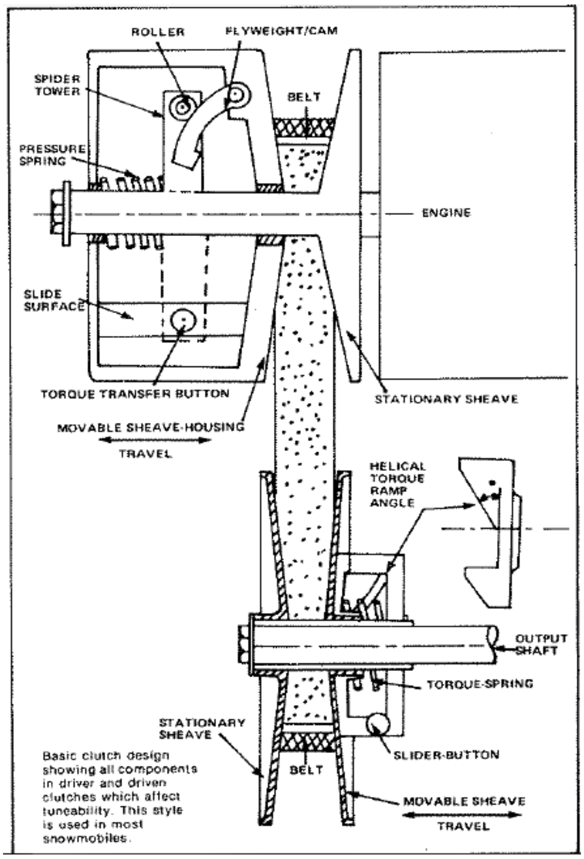


Fig. 1.3

At the driven clutch, another torsion spring presses the movable sheave towards the belt initially. When the centrifugal force developed is greater than the combined force of both these springs, the movable sheave in the driven clutch moves outwards due to belt side force, and the sheave in the driving clutch moves further inside, thus changing the gear ratio.

3.2.1 CVT Ratios:

Since a CVT automatically adjusts itself according to the load conditions, it has only two definite velocity ratios, the lower gear ratio and the higher gear ratio. Between the two ratios, the CVT adjusts itself to infinite number of gear ratios. The diagram shows the CVT running at its lower gear ratio, higher gear ratio and an intermediate ratio.

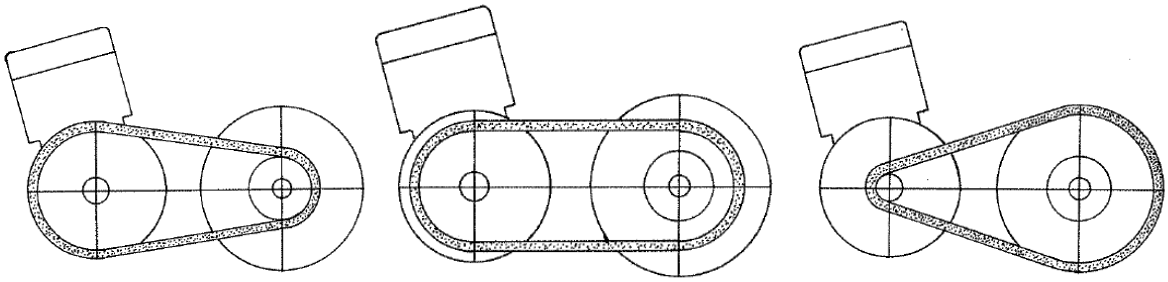


Fig. 1.4

3.2.2 Components:

Driving Clutch/ Primary clutch - The primary clutch consists of the pressure spring and the flyweight mechanism.

- Pressure spring: The pressure spring plays part in the engagement speed of the CVT. It also plays a part in shifting from lower to higher gear ratio. For setting the engagement speed, the spring has to be given a pretension while being installed in the clutch. The pretension is controlled by the amount the spring is compressed while installing in the clutch. For example, a spring with a 50 lb/inch spring rate needs to be compressed by 2 inches to give 100 lbs of pretension. At the same time, when two springs have the same pretension, but different spring rates, then the spring with a higher rate will have greater spring load at full shift. Care must be taken while selecting springs, that the spring does not reach its solid length before full shift of the sheave takes place.

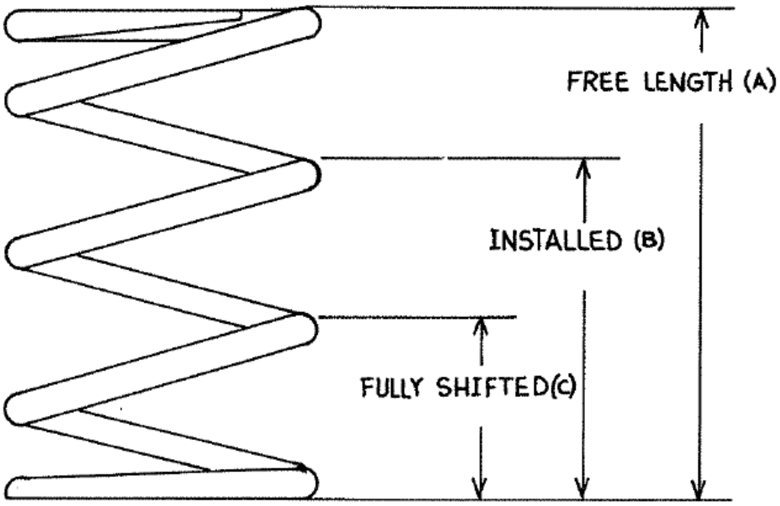


Fig. 1.5

- Flyweight mechanism: This mechanism in the primary clutch controls the engagement as well as shift speed. This mechanism utilizes centrifugal force developed by two or more weights inside the primary clutch to push the sheave towards the belt, against the pretension provided by the pressure spring. There are different types of flyweight mechanisms used in CVTs. Some of them are shown below.

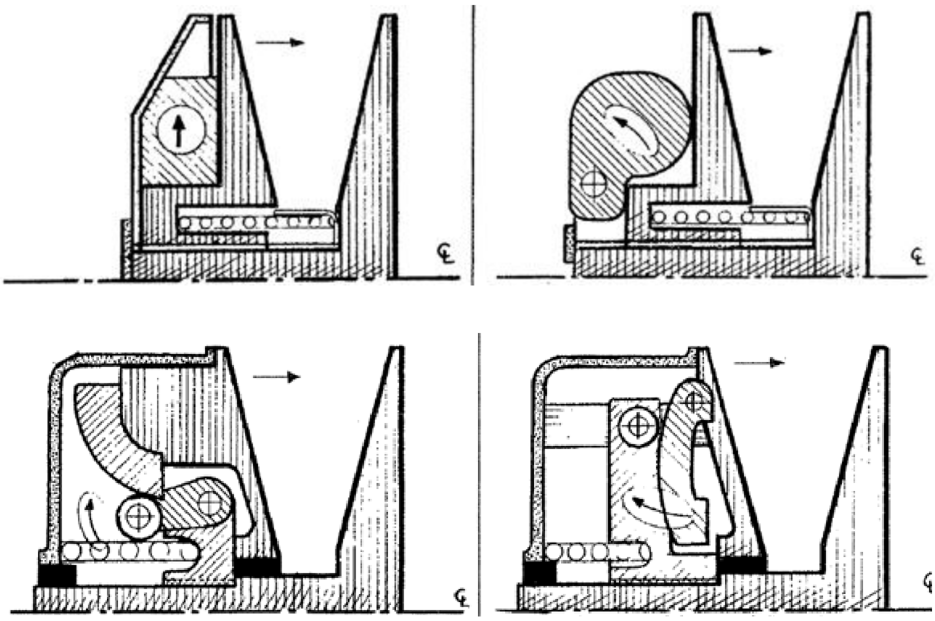


Fig. 1.6

Driven Clutch/Secondary clutch:

- Pretension Spring: The pretension spring in the driven clutch plays a role in initial belt pressure and back shifting. A higher tension gives a quicker backshift. The shift RPM can be set by changing the pretention of this spring, however, it also affects the backshift and efficiency. Holes are provided on the secondary clutch to set the pretension on this spring within a limited range of values.
- Torque Feedback Ramp: The torque feedback ramp consists of three ramps placed around a cylindrical surface working against the sliding buttons in the moving sheave. The angles of the ramps and the radius at which they work both determine the amount of side force that acts on the sheaves. Smaller the ramp angle, greater is the side force produced. A desired combination of ramp angle and pretention spring is used based on the characteristics of the engine and its application.

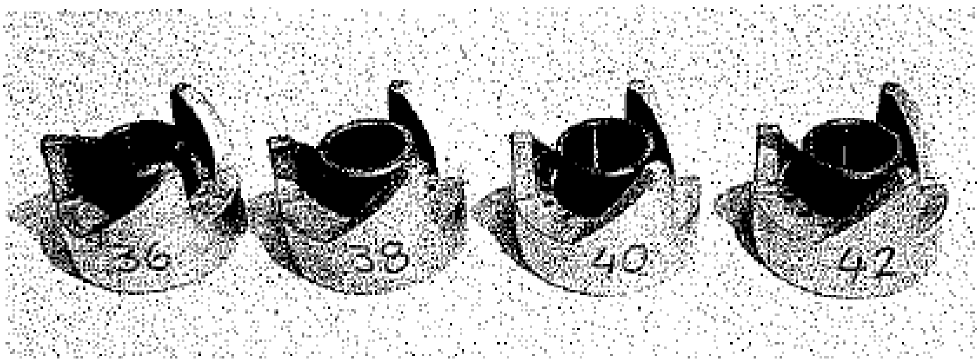


Fig. 1.7

3.2.4 CVTech CVT:

The CVT used in the RedShift Racing prototype is a V-belt driven having 3 sliding cam weights as a centrifugal mechanism. The system gives a lower gear ratio of 3:1 and a higher gear ratio of 0.43:1. It thus provides a ratio range of about 6.97 which is sufficient for its application. The figures below show the exploded view of the primary and secondary clutches of the CVTech CVT.

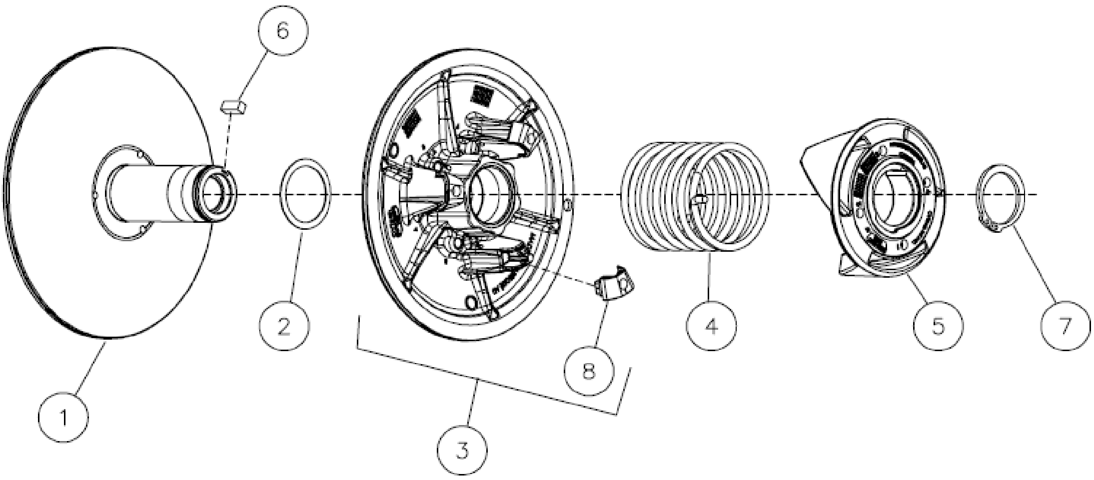


Fig. 1.8

Part No.	Description	Quantity
1	Fixed Flange	1
2	Shim Washer	1
3	Sliding Flange	1
4	Spring	1
5	Cam	1
6	Square Key 1/4”	1
7	Snap Ring	1
8	Cam Shoe	3

Table No. 1.1

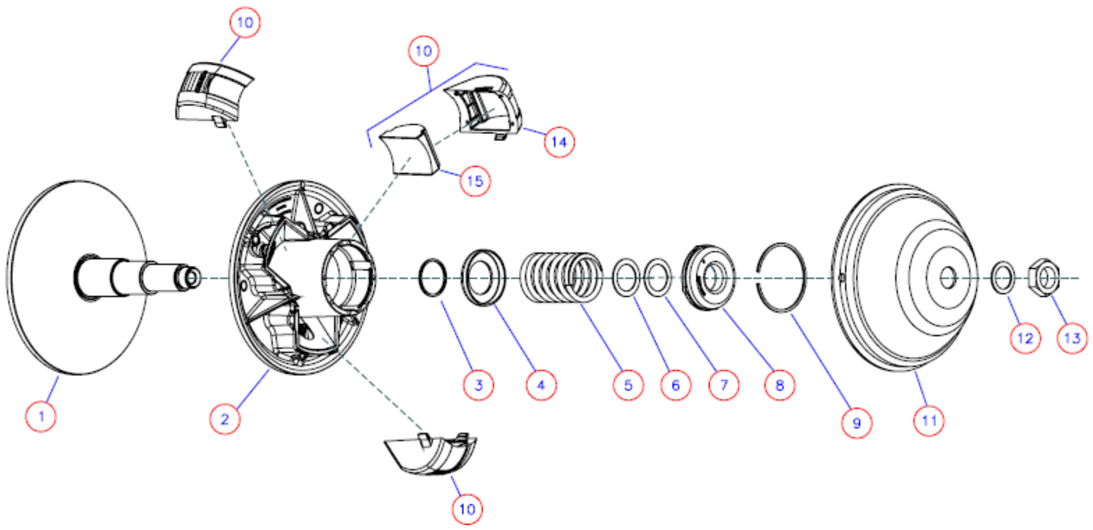


Fig. 1.9

Part No.	Description	Quantity
1	Fixed Flange	1
2	Sliding Flange	1
3	Slider Washer	1
4	Spring Guide Washer	1
5	Spring	1
6	Shim Washer	1
7	Shim Washer	1
8	Spring Cover	1
9	Internal Snap Ring	1
10	Assembly Block	3
11	D1 Cap	1
12	Lock Washer	1
13	Lock Nut	1
14	Block	3
15	Weight 250 gms.	3

Table No. 1.2

3.2 Gearbox

The gearbox is an essential part of any transmission system since it is required to provide the necessary torque and speed output. Gearbox systems can be broadly classified as single speed or multiple speed type. In manual gear systems in automobiles, multiple speed gearboxes are provided to vary the velocity ratios as and when required. But in automatic transmission systems incorporating CVTs, the CVT itself provides variable velocity ratios. Therefore, in such cases, a single speed gearbox can be designed to suite the torque requirements if necessary.

The BAJA SAE vehicle uses a Briggs and Stratton engine having a peak torque output of 18-19 N-m. The CVTech CVT which is to be used in the transmission system gives a lower gear ratio of 3:1. Hence the final torque output will be about 57 N-m. This torque is not sufficient to drive a car having a minimum weight of around 250 Kgs. Hence, a single speed reduction gearbox is essential, in order to provide the required suitable torque. The RedShift Racing prototype utilizes a two stage reduction single speed gearbox.

The gearbox consists of the following components:

- Input Shaft: This shaft is keyed to the driven clutch of the CVT. The first pinion of the gear-train is also mounted on this shaft.
- Compound shaft: This is the intermediate shaft between the input and the output of the gear-train. The gear mating with the first pinion and the second pinion which mates with the output gear.
- The Differential: The differential is required to provide a relative motion between the two driven tires while turning and in rough terrains. It is coupled with the output gear of the gear-train. The output bevel gears of the differential act as the output of the gearbox. The Maruti 800 open differential is used in the customized gearbox, owing to its light weight, small size and low cost.
- Gearbox Housing: The housing not only guards the gear-train, but also provides support to the shafts and bears the axial and radial forces of the gears. The housing is generally manufactured by casting process. It should be rigid and strong, while being light in weight at the same time. The housing also provides an oil inlet and a bleed screw for changing gear oil.
- Oil Seals and Gasket: Oil seals are essential near the input and output shafts to avoid the leakage of gear oil. The gasket performs the function of sealing the gaps between the mating surfaces of the two parts of the housing which are bolted together.

3.3 Rear Axle

Axles are the final drive components along with C.V.Joints & tripods which transmit torque from the differential to the wheel. Since independent suspension system is being used, the axles are fully floating type. This means that no load of the car is taken by the axle, it only transmits the drive. The axles are designed to allow full suspension travel, i.e. they should not restrict the suspension motion. The axles are the part that undergo most loading and thus have to be designed accordingly. The material used for the axle should be very strong and able to undergo shocks. It should also be hard so as to withstand pits and cracks due to cyclic loads. The axles are laid in 3 parts, first from the differential to the first tripod joint, next from the inboard tripod to the outboard CV joint and third from the outboard CV joint to the wheel. Fatigue, whirling, bending and angular deflection are not major design considerations here since as mentioned earlier the axle is fully floating type with light static loads and doesn't require a long life considering the nature of competition.

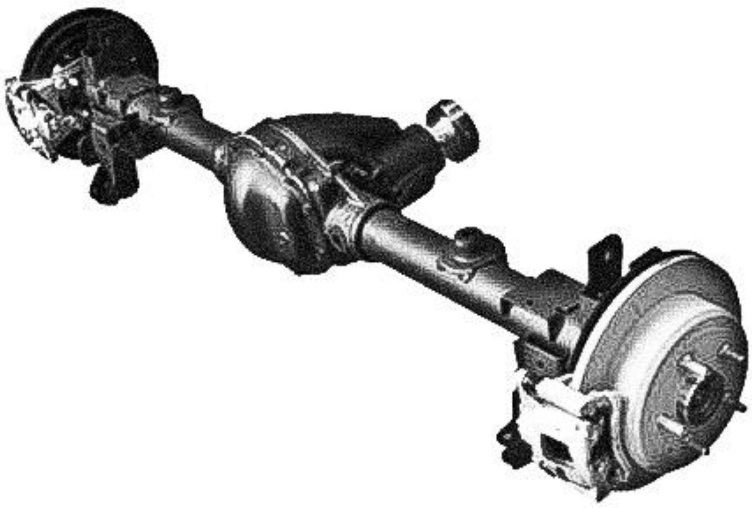


Fig 2.0

Axle joints:

CV Joints allow a drive shaft to transmit power through a variable angle, at constant rotational speed, without an appreciable increase in friction or play. They are mainly used in front wheel drive and all-wheel drive cars. Rear wheel drive cars with independent rear suspension typically use CV joints at the ends of the rear axle half-shafts, and increasingly use them on the propeller shafts. Audi

Quattros use them for all four half-axles and on the front-to-rear driveshaft (propeller shaft) as well, for a total of ten CV joints.

Constant -velocity joints are protected by a rubber boot, a CV gaiter. Cracks and splits in the boot will allow the joint to corrode and a new joint would need to be fitted if the joint is not removed early enough, cleaned, greased, and a new boot fitted.

Rzeppa joints

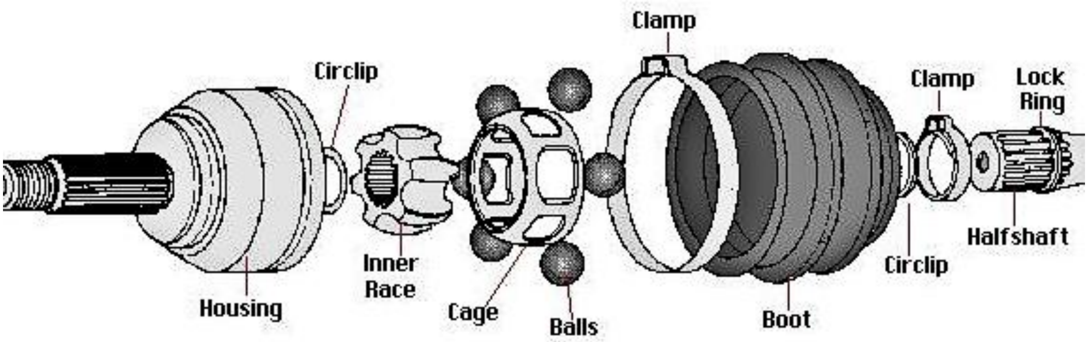


Fig. 2.1

A Rzeppa joint consists of a spherical inner with 6 grooves in it, and a similar enveloping outer shell. Each groove guides one ball. The input shaft fits in the center of a large, steel, star-shaped "gear" that nests inside a circular cage. The cage is spherical but with ends open, and it typically has six openings around the perimeter. This cage and gear fit into a grooved cup that has a splined and threaded shaft attached to it. Six large steel balls sit inside the cup grooves and fit into the cage openings, nestled in the grooves of the star gear. The output shaft on the cup then runs through the wheel bearing and is secured by the axle nut. This joint can accommodate the large changes of angle when the front wheels are turned by the steering system; typical Rzeppa joints allow 45-48 degrees of articulation, while some can give 52 degrees. At the "outboard" end of the driveshaft a slightly different unit is used. The end of the driveshaft is splined and fits into the outer "joint". It is typically held in place by a circlip.

Chapter 4
Fundamental Concepts for Design

This chapter presents the fundamental concepts and formulae on which the designs are based. It gives a summary about the basic parameters used in design. It does not include design details and derivations.

4.1 Gear design concepts and formulae

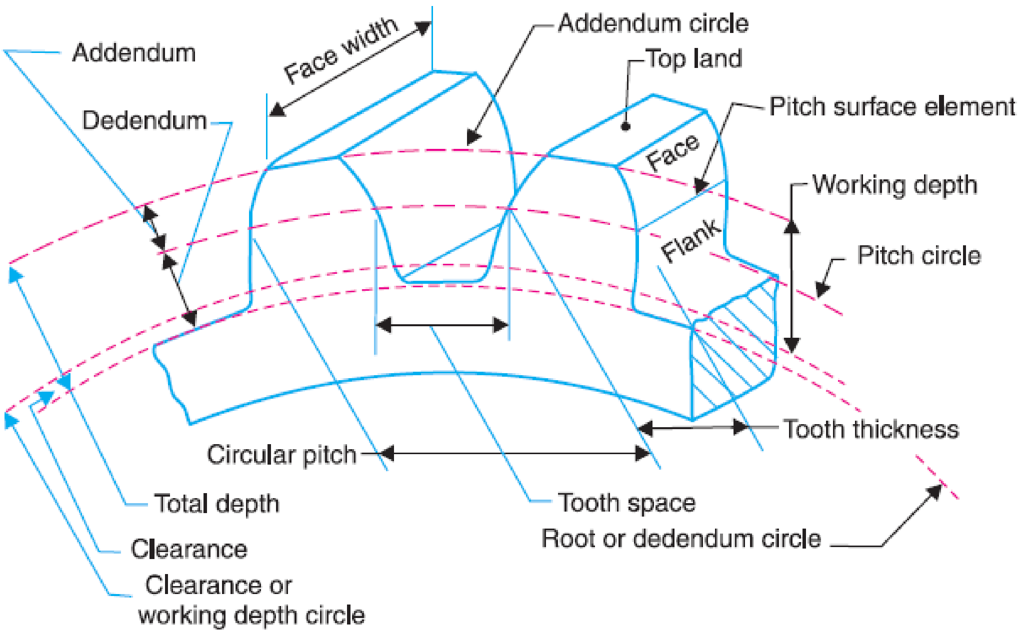


Fig. 2.2

The figure 2.2 shows a view of the standard gear tooth. The following terms are the standard terms used in gear design.

1. Pitch circle. It is an imaginary circle which by pure rolling action, would give the same motion as the actual gear.
2. Pitch circle diameter. It is the diameter of the pitch circle. The size of the gear is usually specified by the pitch circle diameter. It is also known as pitch diameter.
3. Pitch point. It is a common point of contact between two pitch circles.
4. Pitch surface. It is the surface of the rolling discs which the meshing gears have replaced at the pitch circle.
5. Pressure angle or angle of obliquity. It is the angle between the common normal to two gear teeth at the point of contact and the common tangent at the pitch point. It is usually denoted by φ. The standard pressure angles are 14 ° and 20°.

6. Addendum. It is the radial distance of a tooth from the pitch circle to the top of the tooth.
7. Dedendum. It is the radial distance of a tooth from the pitch circle to the bottom of the tooth.
8. Addendum circle. It is the circle drawn through the top of the teeth and is concentric with the pitch circle.
9. Dedendum circle. It is the circle drawn through the bottom of the teeth. It is also called root circle.
10. Circular pitch. It is the distance measured on the circumference of the pitch circle from a point of one tooth to the corresponding point on the next tooth. It is usually denoted by pc.
    Mathematically,
    Circular pitch, $p_c = \pi D/T$
    where D = Diameter of the pitch circle, and
    T = Number of teeth on the wheel.
    The two gears will mesh together correctly, if the two wheels have the same circular pitch.
11. Module. It is the ratio of the pitch circle diameter in millimeters to the number of teeth. It is usually denoted by m. Mathematically,
    Module, $m = D/T$
    The recommended series of modules in Indian Standard are 1, 1.25, 1.5, 2, 2.5, 3, 4, 5, 6, 8, 10, 12, 16
12. Clearance. It is the radial distance from the top of the tooth to the bottom of the tooth, in a meshing gear. A circle passing through the top of the meshing gear is known as clearance circle.

The following four systems of gear teeth are commonly used in practice:

1. 14 1/2° Composite system
2. 14 1/2 ° Full depth involute system
3. 20° Full depth involute system
4. 20° Stub involute system.

The 14 1/2 ° composite system is used for general purpose gears. It is stronger but has no interchangeability. The tooth profile of this system has cycloidal curves at the top and bottom and involute curve at the middle portion. The teeth are produced by formed milling cutters or hobs.
The tooth profile of the 14 1/2° full depth involute system was developed for use with gear hobs for spur and helical gears.
The tooth profile of the 20° full depth involute system may be cut by hobs. The increase of the pressure angle from 14 1/2 ° to 20° results in a stronger tooth, because the tooth acting as a beam is wider at the base.
The 20° stub involute system has a strong tooth to take heavy loads.

Compound Gear trains:

When there are more than one gear on a shaft, it is called a compound train of gear. Whenever the distance between the driver and the driven or follower has to be bridged over by intermediate gears and at the same time a great (or much less) speed ratio is required, then the advantage of intermediate gears is intensified by providing compound gears on intermediate shafts.

In this case, each intermediate shaft has two gears rigidly fixed to it so that they may have the same speed. One of these two gears meshes with the driver and the other with the driven or follower attached to the next shaft.

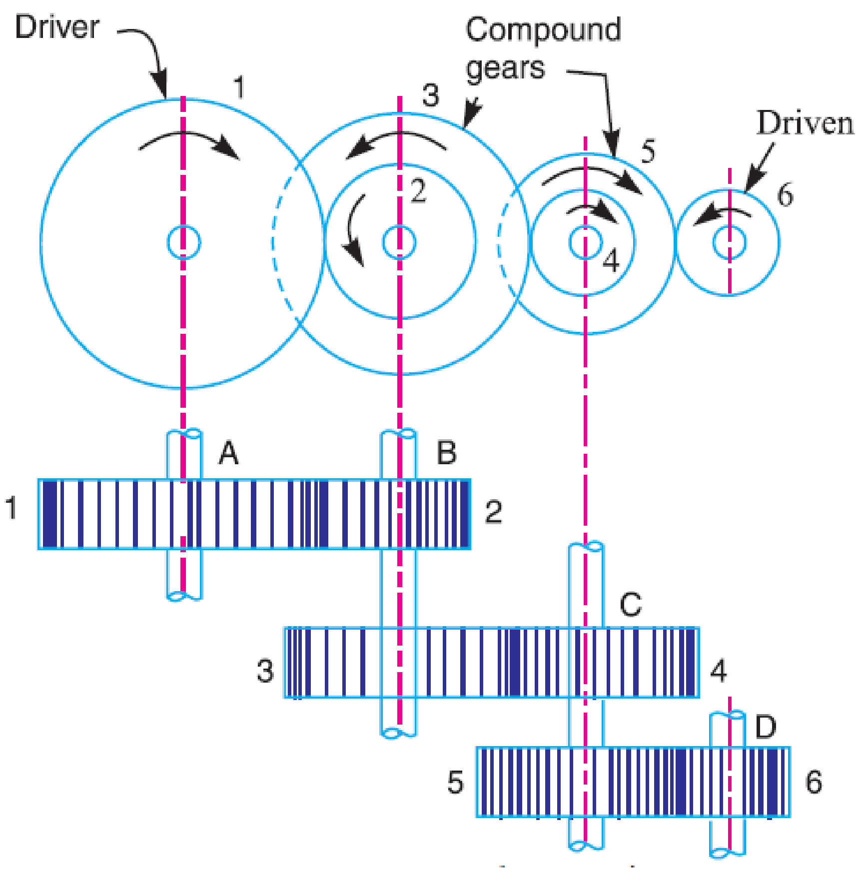


Fig. 2.3

In a compound train of gears, as shown in Fig. 2.3, the gear 1 is the driving gear mounted on shaft A, gears 2 and 3 are compound gears which are mounted on shaft B. The gears 4 and 5 are also compound gears which are mounted on shaft C and the gear 6 is the driven gear mounted on shaft D.

Let N1 = Speed of driving gear 1,

T1 = Number of teeth on driving gear 1,

N2 ,N3 ..., N6 = Speed of respective gears in r.p.m., and

T2 ,T3..., T6 = Number of teeth on respective gears.

Since gear 1 is in mesh with gear 2, therefore its speed ratio is

N1/N2 = T2/T1...(i)

Similarly, for gears 3 and 4, speed ratio is
N3/N4 = T4/T3...(ii)
and for gears 5 and 6, speed ratio is
N5/N6 = T6/T5...(iii)
The speed ratio of compound gear train is obtained by multiplying the equations (i), (ii) and (iii),
(N1/N2) x (N3/N4) x (N5/N6) = (T2/T1) x (T4/T3) x (T6/T5)

Since gears 2 and 3 are mounted on one shaft B, therefore N2 = N3. Similarly gears 4 and 5 are mounted on shaft C, therefore N4 = N5.
The advantage of a compound train over a simple gear train is that a much larger speed reduction from the first shaft to the last shaft can be obtained with small gears. If a simple gear train is used to give a large speed reduction, the last gear has to be very large. Usually for a speed reduction in excess of 7 to 1, a simple train is not used and a compound train or worm gearing is employed.

4.2 Engine Testing

The aim of our experiment is to find natural frequency of the engine (Ꞷn). Natural frequency of the engine is calculated by applying two main fundamentals of forced vibrations viz. Force Transmissibility and Rotating and Reciprocating unbalance.

4.2.1 Calculation of unbalance mass of engine (Mo.e):

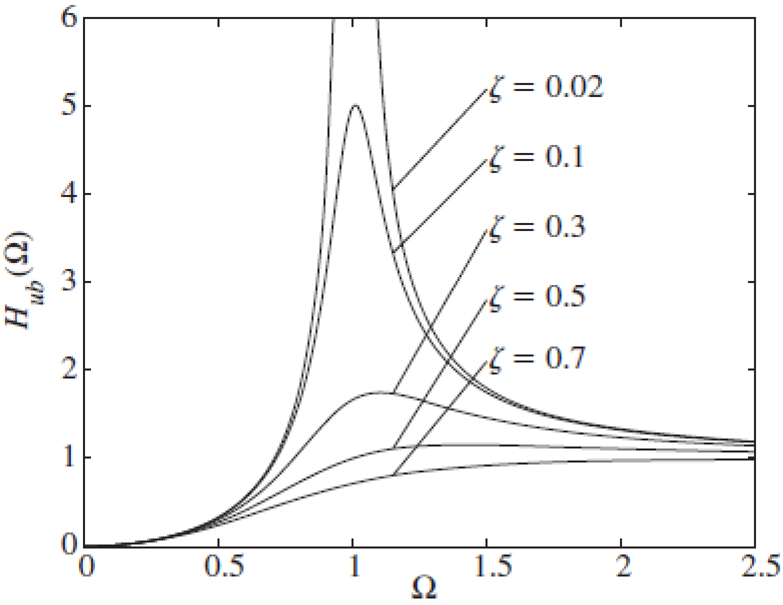


Fig. 2.4

Method 1

From the Response graph of rotating and reciprocating unbalance

We know,

$$\left[\frac{X}{\frac{Mo.e}{M}}\right] = 1$$

@ $r = \infty$

Where r is Frequency ratio

$$r = \frac{\omega}{\omega_n}$$

Therefore in order to make $r = \infty$, we must have $\omega = \infty$. As this is not possible we try to keep ω maximum i.e we tune the engine to run at its rated/ maximum rpm.

By running the engine at its maximum rpm, we can measure the amplitude X in the above equation and mass of the engine M is known.

Thus from above equation we can calculate Mo.e i.e Rotating Unbalance of the engine.

<u>Measuring Amplitude X:</u>

Amplitude X can be measured either by vibrometer or accelerometer.

If we measure X by using Vibrometer we can directly get X.

This is a bit inconvenient method as vibrometer is heavy and bulky and so it is not portable.

We can also measure X by using accelerometer.

The reading obtained is acceleration $\ddot{X}$

$$\frac{\ddot{X}}{(\omega^2)} = X$$

Thus we also need to measure the speed ω at which engine is running. ω can be measured by stroboscope or tachometer.

And from the above equation Amplitude X can be determined.

Method 2

It is seen that practically it is impossible to run engine at maximum rpm therefore reading obtained from the above experiments can be misleading.

Thus another way to find the unbalance of the engine is to use the concept of reciprocating unbalance.

Here we assume that the reciprocating mass (Mo) consisting of the piston and gudgeon pin is concentrated at the end of crank shaft and as the engine rotates it is this mass that causes the unbalance in the engine.

Knowing the stroke length (l) from the engine specification we can calculate the length of crank shaft (e = l/2).

Thus we can find the reciprocating unbalance as Mo.e

4.2.2 Calculation of natural frequency (ƓƆn):

Once the rotating unbalance Mo.e is known Fo at different engine speeds can be calculated as

$$Fo = Mo.e.(W^2)$$

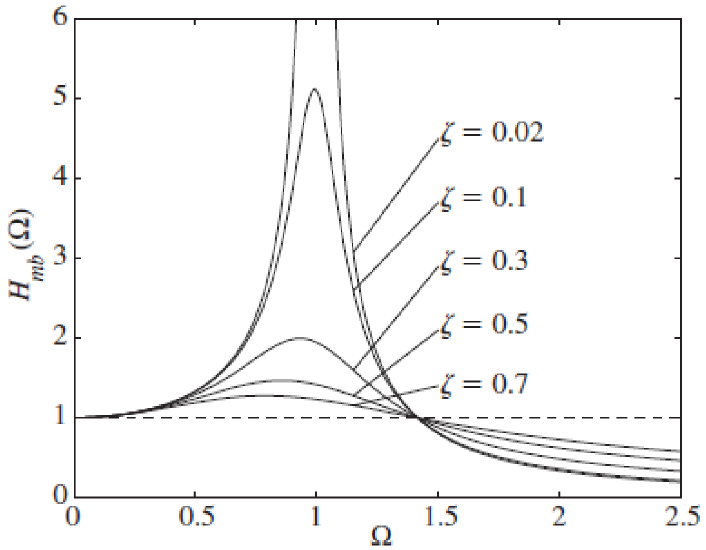


Fig. 2.5

From the graph of force transmissibility,

$\frac{Ftr}{Fo} = 1$ at r = 0 or r = √2

As r = 0 is invalid case for a running engine,

So frequency ratio has to be r = √2

Thus by varying the speed of engine ω we can calculate various values of Fo.

We will try to get that particular ω for which we will get Fo same as Ftr,so that,

$$\frac{Ftr}{Fo} = 1$$

At this condition we know that frequency ratio r = √2

Since $r = \frac{\omega}{\omega n} = \sqrt{2}$

As ω in above equation is known (measured by stroboscope/ tachometer)

Therefore ωn can be calculated.

<u>Measuring Ftr:</u>
Transmitted force Ftr can be calculated by various gauges.

Chapter 5
Implementation of Gearbox Design

This chapter presents the implementation of the literature and concepts of design in the design and analysis of the gearbox. It involves calculations from selecting the reduction ratio to determining the modules and housing parameters.

5.1 Selecting the Reduction Ratio

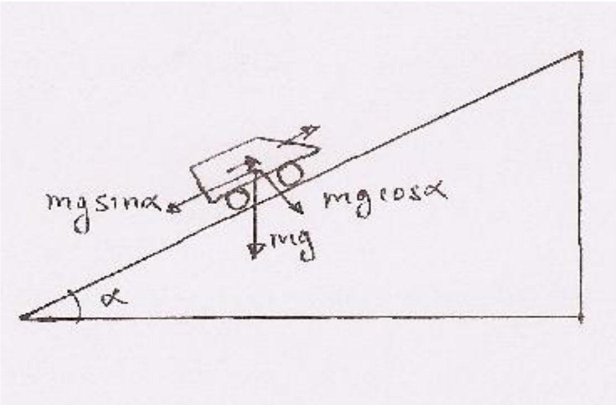


Fig. 2.6

We have designed our system based on the tires selected and the hill climb event in the competition. The below diagram shows the free body diagram of the car moving uphill.

For the tires we have selected,
Coefficient of friction (u)= 0.7
Radius: 0.3125m

Calculation for maximum angle of inclination(a):

- N = mgcos(a)
- uN = mgsin(a)

In the static condition, umgcos(a) = mgsin(a)

From this equation, we get a= 34.99 deg which is approximately 35 deg.
hence, our car can remain in static condition for up to 35 deg inclination.

Let us consider an inclination of 30 deg for the hill climb event. We assume our car weight to be around 360kg for calculations which will be well over the actual weight.

Calculation for Required Torque: (for 30 degree inclination)

- Force pulling the car down-
  P= mgsin(a)
  P=(360 x 9.81 x sin(30))
  P= 1765.8 N
- Torque acting on wheels-
  T=P x r
  T= 1765.8 x 0.3125
  T=551.8125 N-m

Reduction Ratio:

From the engine torque curve (Fig1.2), the engine delivers a maximum torque in the range 19-20 N-m.
Lower Gear Ratio(LGR) = Torque Required/ Engine torque
Lower gear ratio= 551.8125/19 to 551.8125/20
= 29.04 to 27.69
In order to account for the engine efficiency, drive train performance, and other losses, we have considered a factor of safety of 1.5.
Hence, the Lower gear ratio must lie between 41.385 to 43.56
LGR for our selected CVT= 3
HGR for our selected CVT= 0.43

Hence, the final drive train ratio for the gear box = Required reduction/ LGR of CVT
GB reduction ratio= 13.795 to 14.52

Since this ratio is high for single stage reduction, we have designed a two stage reduction gearbox.

Reduction at each stage (i) = square root of GB reduction ratio.

Hence, reduction of each stage must lie between 3.714 and 3.8105

We have selected **17** teeth on the pinion and **64** teeth on the gear at each stage which gives a reduction of **3.7647**, and final drive train ratio of **14.167**.

5.2 Module Selection:

Calculation for module for first reduction (m_n)
We have used mild steel grade EN 24 for the first stage reduction. Since the material for the gear and pinion used is same, the pinion is weaker. We will design the pinion.

Input torque: 60 N-m
Rpm at input (N): 733
Power (P): 4607.648 W

Material Specifications:
Yield strength (Sy): 700 N/mm²
Brinell Hardness no. (BHN): 280

Tooth Strength Calculation:
No. of teeth (z): 17
Reduction (i): 3.7647
Helix angle (b)= 15 deg
FOS= 2

Virtual no. of teeth (z_v) = $z/\cos^3(b)$ = 18.86329
Lewis form factor (y_v) = $0.154 – (0.912/z_v)$ = 0.105652 (from data book)
Bending strength (Sb) = Sy/FOS = 350 N/mm^2
Face width(b) = $10m_n$ (assumption from data book)

Bending strength (Fs) = Sb x b x y_v x π x m_n
= 1169.5614 x m_n^2......(1)

Tooth Load Calculation:
Mean velocity (Vm) = (π x m_n x z x N)/(60,000 x cos(b)) = 0.6758 x m_n
Service factor for intermittent use (about 3 hrs daily) (Cs)= 1
Static Load (Ft) = (P x Cs)/Vm = 6818.064/ m_n
Velocity factor (Cv) = (3 + Vm)/ 3 (for commercially cut teeth)
Dynamic load (Fd) = Ft x Cv = $6818(1 + 0.2253m_n)/m_n$....(2)

Module:
For the design to be safe from strength point of view,
Fs >/= Fd
From equations (1) and (2),
1169.5614 x m_n^2 >/= $6818(1 + 0.2253m_n)/m_n$
Solving the two equations, we get,
m_n = 1.8468mm
So, we have selected a standard module of 2mm.
b= 20mm
Ft = 3409.032 N/mm^2

Checking For Wear Load:
P.C.D for pinion(d) = m_n x z/cos(b) = 35.1994 mm
Ratio factor(Q) = 2i/(1 + i) = 1.5802

Compressive strength(Sc) = (2.8 x BHN) – 70 = 714N/mm^2
Normal pressure angle (a_n) = $\tan^{-1}$ (tan(a) x cos(b)) = 19.37 deg
Modulus of elasticity = 2.06 x 10^5 kgf/cm^2
Load stress factor(k) = (Sc^2 x sin(a_n) x 2) / (1.4 x E) = 1.1725
Wear strength (Fw) = (b x d x Q x k)/$\cos^2$(b) = 1397.98 N/mm^2
For the design to be safe from wear point of view,
Fw>/=Ft
But, Ft is greater than Fw, hence we will have to increase the BHN of the pinion tooth by process of hardening.
Degree of hardening = Root of Ft/Fw = 1.5615
hence we will have to harden the gear to BHN of – 280 x 1.5615 = 437.22

Calculation for module for second reduction (m_n)
We have used mild steel grade EN 36 for the second stage reduction. Since the material for the gear and pinion used is same, the pinion is weaker. We will design the pinion.
Input torque: 225.882 N-m
Rpm at input(N): 194.79
Power(P): 4607.648 W

<u>Material Specifications:</u>
Yield strength(Sy): 900 N/mm^2
Brinell Hardness no. (BHN): 341

<u>Tooth Strength Calculation:</u>
No. of teeth (z) = 17
Reduction (i) = 3.7647
Helix angle (b) = 15 deg
FOS= 2

Virtual no. of teeth (z_v) = z/$\cos^3$(b) = 18.86329
Lewis form factor (y_v) = 0.154 – (0.912/z_v) = 0.105652 (from data book)
Bending strength (Sb) = Sy/FOS = 450 N/mm^2
Face width(b) = 10m_n (assumption from data book)

Bending strength (Fs) = Sb x b x y_v x π x m_n
= 1493.6199 x m_n^2......(1)

<u>Tooth Load Calculation:</u>
Mean velocity (Vm) = (π x m_n x z x N)/(60,000 x cos(b)) = 0.1795 x m_n
Service factor for intermittent use (about 3 hrs daily) (Cs)= 1

Static Load (Ft) = (P x Cs)/Vm = 25669.348/ m_n
Velocity factor (Cv) = (3 + Vm)/ 3 (for commercially cut teeth)
Dynamic load (Fd) = Ft x Cv = 25669.348(1 + 0.059833m_n)/m_n....(2)

Module:
For the design to be safe from strength point of view,
Fs >/= Fd
From equations (1) and (2),
1493.6199 x m_n^2 >/= 26669.358(1 + 0.05983m_n)/m_n
Solving the two equations, we get,
m_n = 2.71344mm
So, we have selected a standard module of 3mm.
b= 30mm we have increased b to 40 mm for better strength.
Ft = 8566.499 N/mm^2

Checking For Wear Load:
P.C.D for pinion(d) = m_n x z/cos(b) = 52.799 mm
Ratio factor(Q) = 2i/(1 + i) = 1.5802
Compressive strength(Sc) = (2.8 x BHN) – 70 = 884.8N/mm^2
Normal pressure angle (a_n) = tan^{-1} (tan(a) x cos(b)) = 19.37 deg
Modulus of elasticity = 2.06 x 10^5 kgf/cm^2
Load stress factor(k) = (Sc^2 x sin(a_n) x 2) / (1.4 x E) = 1.8006
Wear strength (Fw) = (b x d x Q x k)/cos^2(b) = 6440.6164 N/mm^2
For the design to be safe from wear point of view,
Fw>/=Ft
But, Ft is greater than Fw, hence we will have to increase the BHN of the pinion tooth by process of hardening.
Degree of hardening = Root of Ft/Fw = 1.1532
hence we will have to harden the gear to BHN of – 341 x 1.1532 = 393.27

5.3 Shafts and Keys Design :

Input Shaft:
Torque at input (T) = P x 60 / 2 x π x N = 60 x 10^3 N-mm
To account for bending moment, we take T_{eq} = 1.5 x T = 90 x 10^3 N-mm
Allowable sheer stress(Ss) = 0.3 x Sy = 105 N/mm^2
From torsion equation, we get d^3= (16 x T_{eq})/(π x Ss)
Shaft Diameter (d) = 15.379
We will take standard shaft diameter of 20mm.
We have integrated the pinion with the shaft in our design.

Compound Shaft:

Torque at input (T) = P x 60 / 2 x π x N = 225.883 x 10^3 N-mm

To account for bending moment, we take T_{eq} = 1.5 x T = 338.82 x 10^3 N-mm

Allowable sheer stress(Ss) = 0.3 x Sy = 135 N/mm^2

From torsion equation, we get d^3= (16 x T_{eq})/(π x Ss)

Shaft Diameter (d) = 23.38

We will take standard shaft diameter of 30mm.

We have integrated the pinion with the shaft in our design.

The first reduction gear will be solid in construction coupled to the shaft with keys.

ASPECT	PINION 1	GEAR 1	PINION 2	GEAR 2
Number of teeth	17	64	17	64
Material	EN 24	EN 24	En 36	EN 36
FOS	2	2	2	2
Module	2	2	3	3
P.C.D (mm)	35.1994	132.515	52.799	198.77
Addendum dia.	39.1994	136.1994	58.799	204.77
Dedendum dia	30.1994	127.515	45.299	191.27

Table 1.3

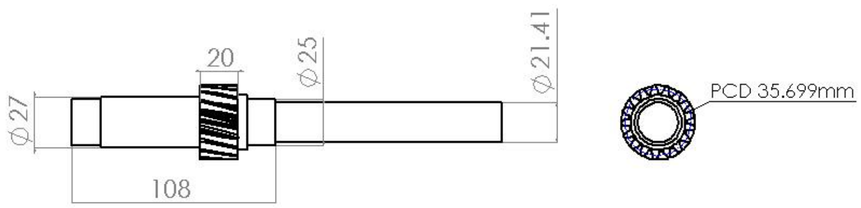


Fig. 2.7

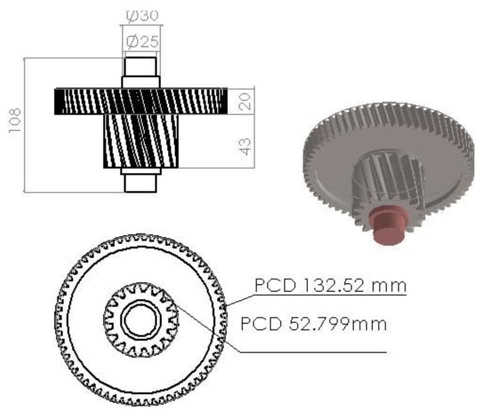


Fig. 2.8

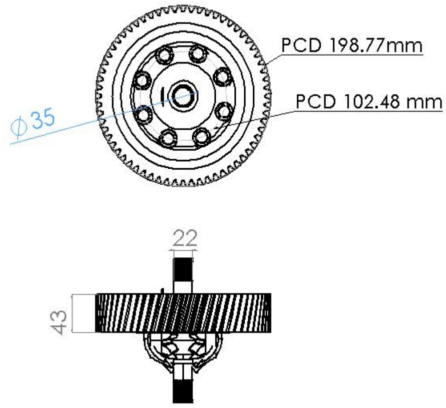


Fig. 2.9

5.4 Housing Design:

According to the Gear diameters, shaft dimensions and bearings selected, we will now design the housing for the gearbox.
We chose to manufacture the gearbox by casting process, using the Aluminum of grade LM6.

Material Specification:
Tensile Strength= 170 N/mm^2
Sheer Strength= 120 N/mm^2

Bearing seat dimensions:
For 32005 X/Q bearing,
Race O.D = 47mm

For 32007 X/Q bearing,
Race O.D = 62 mm

We have kept the casing plate thickness of 7mm with casting point of view. We have done simultaneous analysis on ANSYS along with design to check for failure and stress concentration, since the design of this part is complex for thumb rule calculations.
We have adopted FEA analysis method on the workbench of ANSYS software for the analysis. We have designed the casing with a factor of safety ranging between 6 and 7 for a safe design.

ANSYS report

1. **Geometry:**

Object Name	Gearbox
State	Fully Defined
Definition	
Length Unit	Millimeters
Element Control	Program Controlled
Bounding Box	
Length X	0.178 m
Length Y	0.33332 m
Length Z	0.24458 m

Table 1.4

2. Assigning Material Properties:

Material		
Assignment	Aluminum Alloy LM 6 Series	
Properties		
Volume	1.2939e-003 m³	9.5056e-004 m³
Mass	3.4288 kg	2.519 kg

Density	2650 kg m^-3
Coefficient of Thermal Expansion	2.e-005 C^-1
Specific Heat	875 J kg^-1 C^-1

Young's Modulus Pa	Poisson's Ratio	Bulk Modulus Pa	Shear Modulus Pa	Compressive Yield Strength Pa
7.1e+010	0.33	6.9608e+010	2.6692e+010	1.7e+008

Tensile Ultimate Strength Pa	Tensile Yield Strength Pa
3.1e+008	1.7e+008

Table 1.5

3. Meshing:

Object Name	*Gearbox Mesh*
State	Solved
Physics Preference	Mechanical
Sizing	
Element Size	1.e-003 m
Minimum Edge Length	1.7908e-004 m
Triangle Surface Mesh	Program Controlled
Advanced	
Shape Checking	Standard Mechanical
Element Mid-side Nodes	Program Controlled
Straight Sided Elements	No
Number of Retries	Default (4)
Extra Retries For Assembly	Yes
Mesh Morphing	Disabled
Statistics	
Nodes	2056607
Elements	1252306

Table 1.6

4. Applying Boundary Conditions:

Object Name	*Gearbox*
State	Solved
Definition	
Physics Type	Structural
Analysis Type	Static Structural 3D
Options	
Environment Temperature	22. °C

Object Name	*Force*	*Force 2*	*Force 3*	*Force 4*	*Force 5*
State	Fully Defined				
Scope					
Scoping Method	Geometry Selection				
Geometry	1 Face				
Definition					
Type	Force				
Define By	Components			Vector	
Coordinate System	Global Coordinate System				
X Component	428.12 N (ramped)	646.38 N (ramped)	-1074.5 N (ramped)		
Y Component	0. N (ramped)				
Z Component	0. N (ramped)				
Suppressed	No				
Magnitude				138.4 N (ramped)	771.74 N (ramped)
Direction				Defined	

Object Name	*Force 6*	*Force 7*	*Force 8*	*Fixed Support*	*Fixed Support 2*
State	Fully Defined				Suppressed
Scope					
Scoping Method	Geometry Selection				
Geometry	2 Faces	1 Face		4 Faces	
Definition					
Type	Force			Fixed Support	

Define By	Vector			
Magnitude	821.93 N (ramped)	463.63 N (ramped)	569.06 N (ramped)	
Direction	Defined			
Suppressed	No			Yes

Object Name	*Fixed Support 3*	*Fixed Support 4*	*Fixed Support 5*	*Fixed Support 6*	*Fixed Support 7*
State	Suppressed				
Scope					
Scoping Method	Geometry Selection				
Geometry	4 Faces	No Selection	4 Faces	2 Faces	4 Faces
Definition					
Type	Fixed Support				
Suppressed	Yes				

Object Name	*Fixed Support 8*	*Fixed Support 9*
State	Fully Defined	
Scope		
Scoping Method	Geometry Selection	
Geometry	1 Face	
Definition		
Type	Fixed Support	
Suppressed	No	

Table 1.7

5. Solutions:

Object Name	*Gearbox*
State	Solved
Adaptive Mesh Refinement	
Max Refinement Loops	1.
Refinement Depth	2.
Information	
Status	Done
Object Name	Solution Information
State	Solved

Solution Information	
Solution Output	Solver Output
Display Points	All
FE Connection Visibility	
Activate Visibility	Yes
Display	All FE Connectors
Draw Connections Attached To	All Nodes

Object Name	*Equivalent Stress*	*Total Deformation*	*Shear Stress*
State	Solved		
Scope			
Scoping Method	Geometry Selection		
Geometry	All Bodies		
Definition			
Type	Equivalent (von-Mises) Stress	Total Deformation	Shear Stress
By	Time		
Suppressed	No		
Orientation			XY Plane
Coordinate System			Global Coordinate System
Integration Point Results			
Display Option	Averaged		Averaged
Results			
Minimum	449.2 Pa	0. m	-3.7084e+006 Pa
Maximum	1.2449e+007 Pa	1.1033e-005 m	3.45e+006 Pa
Minimum Occurs On	Solid		
Maximum Occurs On	Solid		

Object Name	*Safety Factor*	*Life*
State	Solved	
Scope		
Scoping Method	Geometry Selection	
Geometry	All Bodies	
Definition		
Design Life	1.e+009 cycles	
Type	Safety Factor	Life
Suppressed	No	
Results		
Minimum	6.6465	1.e+008 cycles
Minimum Occurs On	Solid	

Table 1.8

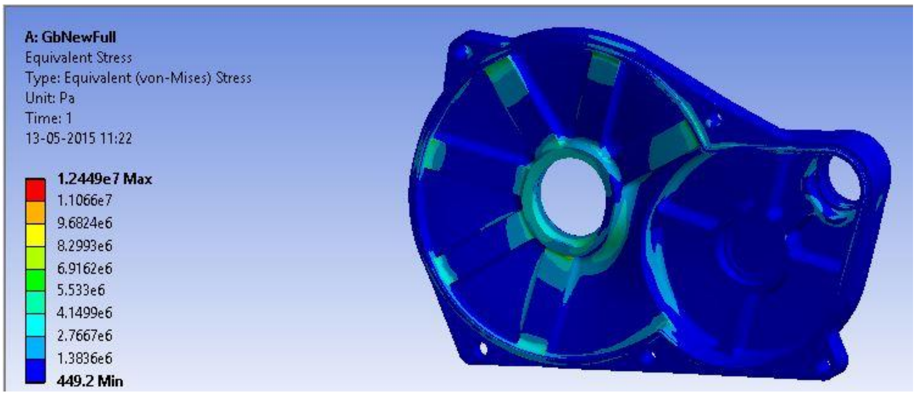


Fig 3.0

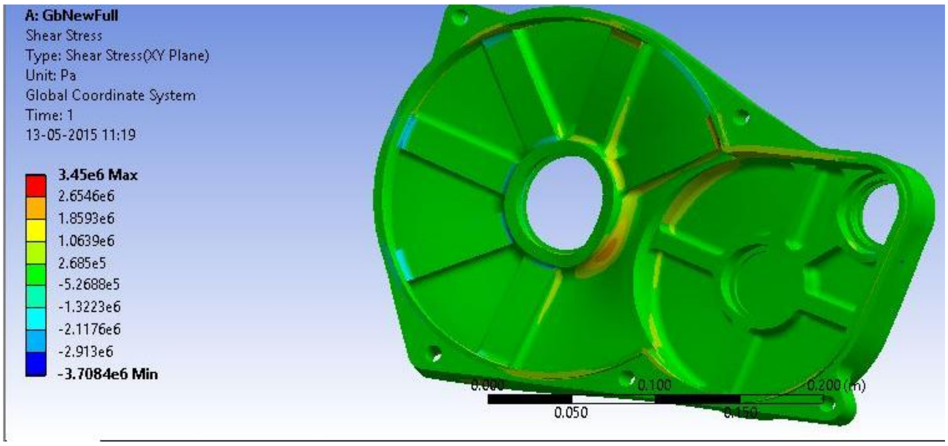


Fig 3.1

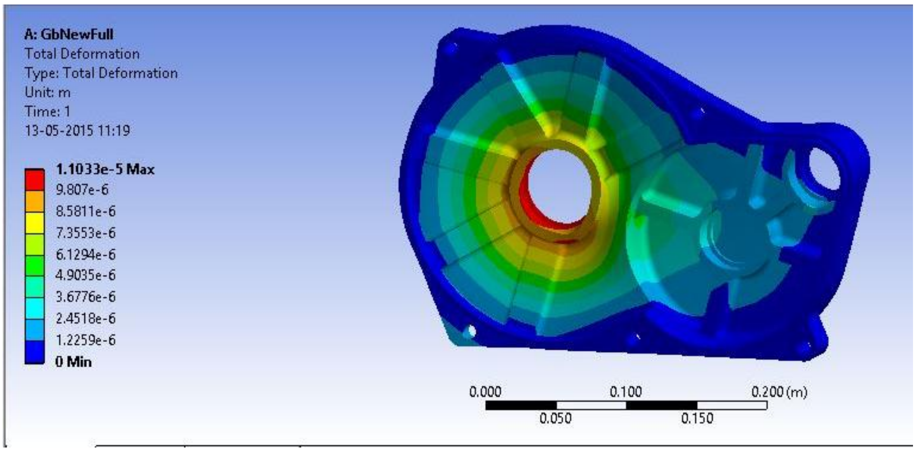


Fig 2.2

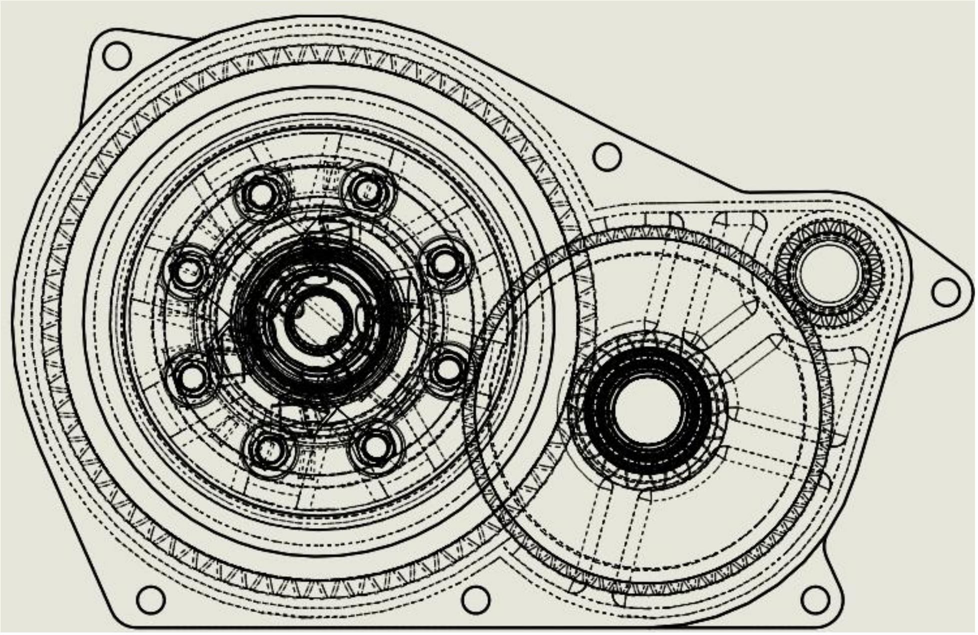


Fig. 3.3

Fig. 3.4

Chapter 6
CVT tuning

This chapter presents the application of clutch tuning concepts to effective tuning of the CVT for better performance and acceleration. It involves the study of CVT characteristics over a range of permutations to arrive at an optimum result.

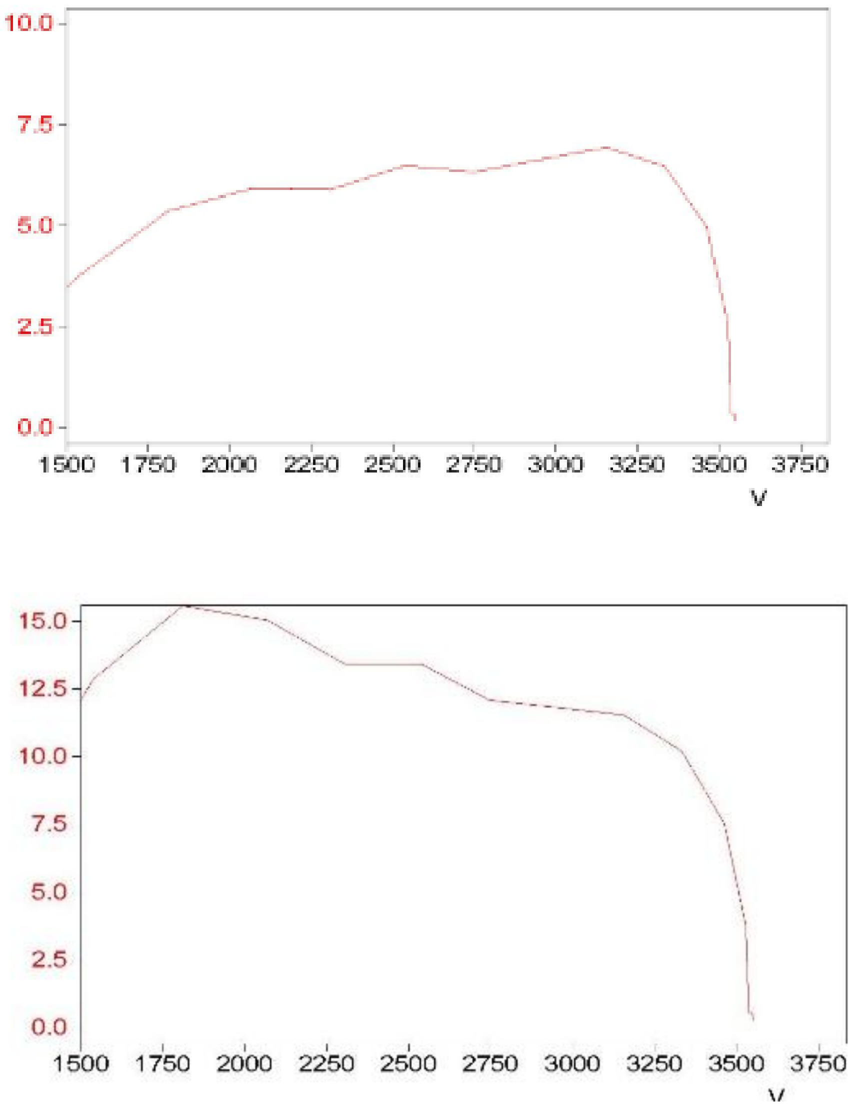


Fig. 3.5

The figure shows the torque and power variation graphs for the briggs and stratton 10 HP engine. The CVT must be tuned in such a way that the engagement zone is a few rpm before the maximum torque delivered by the engine, and the shift zone is a few rpm before the maximum power delivered by the engine, so that we can keep the engine running at maximum power with an increasing speed until the shift stops.
The graphs show that the engagement must be in the range of 1800 to 2200 rpm and the shift must be in the range of 1900 to 3300 rpm. We have thus selected our engagement and shift ranges.

The CVTech CVT which has the following tuning specifications:

- Flyweights: 250gm/200gm
- Spring rate(k): 15N/mm
- Radius of rotation for flyweights: 31.5 mm
- Shift ratio: 6.9769
- Ramp Radius: 34 mm
- Ramp angle: 36 deg

We arrived at a satisfactory result by varying the spring rate and pretension of the driver spring. We could also have arrived at a better result by changing the weights, but changing the weights requires close accuracy so as to not change the center of gravity of the flyweights. Hence, we chose to manipulate the spring specifications, which is an easier option.

<u>Basic formulae regarding engagement and shift speeds</u>

Engagement equation:

$kx = 3mrw^2$

where,
k=spring rate
x=pretension
m=mass of 1 flyweight
r=radius of rotation
w=angular speed

Shift equation:

$SF + kx = mrw^2$

Where, SF is the side force which is given by equation:

SF = (T x Shift ratio x 12)/ (ramp dia x tan(a))

where,
T= torque
a= ramp angle

Tuning variation grid: (for m=250)

X(mm)	k(N/mm)	Engagement rpm	Shift rpm
44.09	5.87	2000	2505.86
44.09	8	2333.62	2899
32.09	13	1537.88	2953.7
36.09	8	2111.319	3452.8
15.587	15	1900	2396
44.09	15	2195.44	3534.7
32.09	15	1891.04	3290.7
44.09	19	1798	3020
44.09	20	1844	3032

Table 1.9

From the table,
We have selected 250gm flyweight with 19N/mm spring having pretention of 44.09 mm. We have manufactured the suitable spring.

Chapter 7
Engine damping design

This chapter presents the experimentation on engine vibrations to find out the natural frequency of the engine and propose a suitable damper and spring. It involves calculation of the Rotating unbalance of the engine and the use of experiment results to determine natural frequency.

A experiment had been conducted to calculate the natural frequency of engine once the unbalance mass of the engine was determined.

A digital rpm meter was used to record the rpm of the engine for various iterations in order to achieve desired condition.

A vibration transducer was used to determine the amplitude of vibrations (X) of the mounts from which the force transmitted from the engine was calculated. The amplitude measured was in terms of microns.

Firstly the excitation frequency of the engine (Fo) was calculated for a particular speed using rpm meter. Then the amplitude of the mounts was determined from vibration transducer and transmitted force (Ftr) was calculated for the same speed.

This procedure was repeated at different speeds till same values of Fo and Ftr were obtained which was the desired condition and the speed (ω) was recorded.

<u>Calculation of Unbalanced Mass (Mo.e):</u>

Since calculations of unbalance mass using method 1 did not give us expected results we used method 2 for calculation of unbalance mass.

From Engine specifications,

Bore Diameter (D) = 3.23in = 82.042mm

Stroke Length (L) = 2.28in = 57.912mm

Therefore Length of crank shaft (e) = $L/_2 = 57.912/_2 = 28.956mm$

Mass of reciprocating parts (Mo):-

Mass of piston = 0.223 kg

Mass of gudgeon pin = 0.368 kg

Therefore Mass of reciprocating parts = Mass of piston + Mass of gudgeon pin

= 0.223 + 0.368

Therefore Mo = 0.591 kg

Reciprocating unbalance = Mo.e = 0.591*28.956

= 0.0171 kg-m

Thus from unbalance mass Mo.e, Excitation force Fo at various speeds can be calculated.

Determination of natural frequency (ῼn):-

Referring to the graph of force transmissiblity,

The conditon of $\frac{Ftr}{Fo}$ = 1 is obtained by trial and error method.

This condition is achieved at speed ῼ = 2835 rpm

At this conditon,

Frequency ratio r = ῼ/ ῼn= √2

i.e 2835/   = √2

Therefore ῼn = 2004.64 rpm

=209.92 r/s

Conclusion:

Since the natural frequency of the system is 2004.64 rpm, resonance will occur at this point. If the engine is not damped at this stage, it may damage the mounts and cause power losses, which affects the overall working of the drive train. Therefore, the damping factor and damping coefficient of the mounts should be selected such that the engine is in isolation zone and should be able to sustain resonance condition.

Chapter 8
Conclusion and Scope

Through the study of previous designs, we came to the conclusion that the implementation of a differential is imperative, for the required maneuverability. Material research and property wise analysis helped us select Lm-6 series aluminum for the gearbox housing and EN-24 and EN-36 series for the gears. Through revised designs, we have achieved an FOS of a minimum of 6.64 at maximum stressed section. Permutations for CVT engagement and shift speed calculations provided us optimum values for the tuning parameters like flyweight values, spring loading and stiffness.

After rigorous testing and enduring for about 70kms run, the entire transmission system is performing consistently with satisfactory results. It has sustained all types of rough and inclined gradients successfully.

With suitable modifications the ATV has scope for application in various sectors such as military vehicle, for recreational sports and agricultural ploughing. They can also during natural calamities such as earthquakes to reach people who are inaccessible.

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Last but not the least, we would like to thank our seniors and colleagues for their valuable inputs in this project.

BE Project 2014 -15

Design and Analysis of the Drive Train of an All-Terrain Vehicle

1. Priyank Metha 2. Rohit Salgaonkar 3.Soham Gokhale
Guide: Prof. V S Narwane

Group No. 48

Abstract

Drivetrain basically concerns with conveying power from the vehicle's engine, through the transmission to the drive wheels on the vehicle to control the amount of torque. It comprises of the Engine, CVT, Gearbox, Differential, CV joints and Axle shafts. The ATVs are meant for operation on rough, off-road terrains which demand higher torque and a smooth transmission. An automatic transmission system through the use of a CVT has been incorporated due to its advantage of having smooth operation (infinite gear ratios) and compactness over manual transmission. Also CVT tuning enables us to operate the engine at power peak so that power is transmitted most efficiently. A self-designed two stage reduction gearbox is coupled with the CVT to achieve the desired torques requirements for the terrain. An open differential is integrated with the gear train to provide aid in steering. While designing and analysis emphasis is laid on, material selection, selection of proper FOS, method of production for gears and the gearbox so as to ensure longer life for the system, minimal transmission losses and also minimize the total cost of the system so that it can compete in the existing market. Also proper system layout has been ensured for ease in serviceability.

Introduction

Our project focuses on the study, design, analysis and manufacturing of the power transmission system of the BAJA all-terrain vehicle in association with team *Redshift Racing.* The project encompasses the study of the previous gearbox designs of the team, the Continuously Variable Transmission System (CVT), and the development of a new design for a better performance of the vehicle.

We focus on the optimization of the current design in terms of the cost of manufacture and the overall weight of the system, along with the improvement in the performance characteristics like acceleration, maneuverability and speed. We design the components on CATIA software while carrying out structural analysis on ANSYS software. The project involves the design of the entire two stage reduction gearbox, study of the performance characteristics of the CVT and its suitable modification with rigorous testing, design and manufacture of the rear-axle, and stability and mounting considerations of the entire system in particular.

The system we design has applications in other fields like agriculture, snow-mobiles, and ATV sports. CVT systems are an emerging trend in today's automobile industry. Due to its low cost, adaptability and compact assembly, CVT systems are suitable for applications in tractors and other agricultural equipment. The system we design implementing the CVT with low cost components such as the Briggs and Stratton Engine and gearbox considering mass production, is suited for these applications. Since it eliminates the use of a clutch and the need to switch gears, we believe it will revolutionize the industry in terms of such application in the near future.

Literature Review

Clutch Tuning:

Engagement speed: The engagement speed of the CVT is determined by the driver pressure spring and the flyweights.

Shift speed: The shift speed is the speed of the engine where the velocity ratio shifts from the lower ratio to the higher ratio. It is determined by the driver settings and the torsion spring and ramp angles on the driven.

The CVT should be tuned such that the engine runs in the power peak range under its entire range of operation. This can be achieved by varying the parameters like the spring constant, spring pretension, torsional stiffness of the spring, flyweight masses and ramp angle.

Gearbox:

For a reduction ratio of above 4, a multiple stage compound gear train is recommended. A two stage drive train is suited for the reduction ratio of about 14, with about 3.7 reduction ratio at each stage. Helical gears perform well in high speed operations where smooth transmission and low noise is essential. Helical gears are therefore suited for vehicle gear box systems. Full depth involute system is the most commonly used system for the gear profile suited for high speed transmissions.

Uniform teeth profile, load application on a single tooth at a time, neglecting the radial and compressive forces, are some of the assumptions made during the gear calculations. To account for the neglected factors, a suitable factor of safety of 2.5 is selected for the selection of the modules.

Selection of bearings is based on the factors like axial loads, radial loads, average speed of operation and life of the bearing.

Shaft design is based on the torsional equations. Length of shaft is determined considering limited elastic deformation during operation.

The housing design constitutes space for the gears, and seats for the bearings. The casing should sustain forces like weight of the gears, radial and axial reactions of the bearings, and any outside impact to protect the components from damage. Housing design is based on basic principles of Machine Design and accepted theories of failure in tensile, bending and shear modes.

Material Selection:

Materials are selected based on their properties, availability, cost and suitability for the required application. Some examples of properties considered for material selection are yield strength, ultimate tensile strength, hardness, ability for heat treatment, machinability, allowable working temperature.

Methodology/ Research work

1. Study of the current design. Analysis of its drawbacks and plus points.
2. Plotting of the necessary improvements/changes.
3. Market research for materials, manufacturing processes, cost requirements.
4. Finalizing the project plan and cost estimation.
5. Theoretical calculations and CAD.
6. Analysis of component designs, finalizing of designs.
7. Manufacturing processes, purchase of supplemental parts.
8. Assembly of the components.
9. Testing of the assembly.
10. Experimentation and measurement of performance parameters.
11. CVT tuning.
12. Engine vibrational analysis.

Our current research work has three main subdivisions:

- Design and analysis of components including the gears, rear axle, housing, etc.
- Study of performance characteristics of the CVT and parameters that determine them. Study of CVT tuning and implementation.
- Study of Vibrational analysis of an engine. Experimental analysis of the current engine and suitable damping and mounting of the engine.

For the reduction gearbox, the following research work has being undertaken:

- Gear design and material selection. Gear manufacturing processes, their reliability, cost optimization. Hardening processes.
- Housing design. Materials properties required for sand casting. Design considerations for sand casting. Determination of tolerances for bearing seats, shaft designs. CNC machining for critical sections.
- Bearing and oil seals selection.
- System layout considerations for proper serviceability.
- FEA method for structural analysis.

For CVT systems:

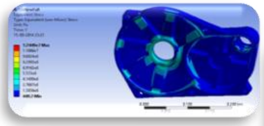
- Understanding the working of the CVT systems. Forces and basic physics of the working.
- Study and determination of working parameters such as engagement RPM, shift RPM, etc.
- Components determining parameters for design such as properties of springs, torsional springs, flyweight design and manufacture, ramp angles for shift, etc.
- Modification of existing system through reverse engineering for suitable performance.

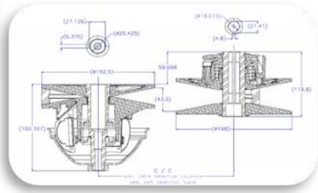
Engine Mounting:

- Study of Vibrational analysis and vibration damping.
- Experimentation for vibration characteristics of the engine.
- Selection of suitable damping supports for reduction in transmitted vibrations.
- Nodal Analysis using ANSYS.

Results and Conclusion

- Through the study of previous designs, we came to the conclusion that the implementation of a differential is imperative, for the required maneuverability.
- Material research and property wise analysis helped us select Lm-6 series aluminum for the gearbox housing and EN-24 and EN-36 series for the gears.
- Through revised designs, we have achieved an FOS of a minimum of 6.64 at maximum stressed section.
- Permutations for CVT engagement and shift speed calculations provided us optimum values for the tuning parameters like flyweight values, spring loading and stiffness.
- After rigorous testing and enduring for about 70kms run, the entire transmission system is performing consistently with satisfactory results.
- It has sustained all types of rough and inclined gradients successfully.





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Baja 2014 scores

Position	Team	Design Report (80)	Cost Report (50)	Sales (20)	Static Design (150)	Acceleration (65)	Top Speed (55)	Hill Climb (50)	Skid Pull (50)	Handling (80)	Endurance Race (400)	Penalties	Total (1000)
1	CUT 8	64.416	0	8.462	**132.644**	26.809	**55**	44.522	31.413	**80**	344.828	N/A	**788.094**
2	NMMU 7	73.766	41.892	11.538	123.993	21.52	16.194	32.006	33.125	62.376	317.241	N/A	733.651
3	TUKS 2	76.883	**50**	13.846	132.536	26.803	40	29.793	**50**	64.754	227.586	N/A	712.201
4	TUT 1	0	32.095	6.923	126.254	5	36.866	**50**	N/A	54.881	**400**	N/A	712.019
5	TUT 3	63.377	29.392	8.462	114.989	25.532	11.03	44.554	50	62.964	186.207	N/A	596.507
6	CUT 10	0	0	9.231	115.523	48.756	10.448	5	7.201	43.114	158.621	N/A	397.894
7	TUKS 5	**80**	40.878	9.231	116.079	37.538	19.552	15.78	5	N/A	27.586	N/A	351.644
8	UKZN 6	0	12.162	9.231	106.445	**65**	5	N/A	N/A	5	110.345	N/A	313.183
9	WITS 14	44.676	24.324	6.154	112.581	N/A	N/A	N/A	N/A	N/A	55.172	N/A	242.907
10	SCE 18	61.299	8.108	**20**	126.154	N/A	N/A	N/A	N/A	N/A	0	N/A	215.561
11	VUT 11	0	45.27	0	93.323	N/A	N/A	N/A	N/A	N/A	75.862	N/A	214.455
12	VUT 12	11.428	43.919	5.385	77.209	N/A	N/A	N/A	N/A	N/A	0	N/A	137.941
13	WITS 13	51.948	24.324	N/A	N/A	N/A	N/A	N/A	N/A	N/A	0	N/A	76.272
14	UKZN 9	0	12.162	0	0	N/A	N/A	N/A	N/A	N/A	0	N/A	12.162
15	NWU 4	N/A	N/A	N/A	N/A	N/A	N/A	N/A	N/A	N/A	N/A	N/A	0